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Testing of Face-Milled Spiral Bevel Gears at High-Speed and Load

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March 2001

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Abstract: Spiral bevel gears are an important drive system components of rotorcraft (helicopters) currently in use. In this application the spiral bevel gears are required to transmit very high torque at high rotational speed. Available experimental data on the operational characteristics for thermal and structural behavior is relatively small in comparison to that found for parallel axis gears. An ongoing test program has been in place at NASA Glenn Research Center over the last ten years to investigate their operational behavior at operating conditions found in aerospace applications. This paper will summarize the results of the tests conducted on face-milled spiral bevel gears. The data from the pinion member (temperature and stress) were taken at conditions from slow-roll to 14400 rpm and up to 537 kW (720 hp). The results have shown that operating temperature is affected by the location of the lubricating jet with respect to the point it is injected and the operating conditions that are imposed. Also the stress measured from slow-roll to very high rotational speed, at various torque levels, indicated little dynamic affect over the rotational speeds tested.
Keywords: Gears, Transmissions

1 INTRODUCTION

Spiral bevel gears are important components in all current rotorcraft drive systems. Their use in this application is essential since they are used to change rotational direction between the horizontal engines and vertical rotors. Using them in this application typically results in the gear mesh operating at very high rotational speed and at high torque levels. An example of a helicopter main rotor transmission using spiral bevel gears is shown in Fig. 1. In this application spiral bevel gears are used for two purposes[1]. The first purpose is to provide flexibility in engine placement within the rotorcraft. The second purpose is to combine the two engine inputs into a common output gear that in turn drives a planetary gear mesh and drives the main rotor. This very flexible aspect of spiral bevel gears makes their use very attractive to drive system designers. Since drive system designers are not restricted with certain shaft orientations, high reduction ratios with a minimum of available volume can be accomplished.

The operation of drive systems as used in aerospace applications is quite different from low speed or automotive type applications. The gearing system is pressure lubricated using multiple jets at many locations within the gearbox. The lubricant is removed from the gearbox using scavenge pumps to a reservoir. This is done to minimize gear-lubricant churning and provide the drive system with the highest efficiency possible. This important aspect of the drive system, for aviation applications, will minimize the weight of the lubricant coolers and associated components.

For aerospace applications, structurally, the components must be light-weight, have high load capacity, and be very reliable. The gears are made from the highest quality alloys that exhibit high surface hardness to endure the extreme contact loads yet have a tough core strength that can tolerate the teeth deflecting due to the high bending loads. Gears in this application are carburized and finish ground to have components manufactured to very high tolerances. All aspects of the gear design process are important because gears operating in an aerospace environment are pushed to the limits of operation.

The work of others in experimentally measuring the thermal performance and structural behavior of gears has been rather limited. This is due to special equipment necessary to make these difficult measurements. The amount of experimental effort in these areas is depleted further when considering spiral bevel gears. The roots of making gear measurements, however, dates back to the 1930's and the work conducted by Blok[2,3]. In more recent years other researchers have completed studies on parallel axis and bevel gearing investigating the influence of operation and other parameters on the thermal performance[4-15]. The measurement of the structural performance of spiral bevel gears has mainly been investigated using helicopter transmissions or simulations of the aerospace requirements. A description of some of the work available in the open literature is contained in Refs. 16-19.

The objective of this work contained in this paper is to summarize the experimental results attained in the NASA Glenn Research Center Spiral Bevel Gear Test Facility. The components used in this facility are representative of aerospace quality gears and levels of speed and load are also representative of the high speed and load conditions that these components are required to operate under. The testing has been done for thermal and structural behavior. Tests were run with an instrumented pinion member. Conditions of the tests were run from low to high speed (up to 14400 rpm) and up to 537 kW (720 hp).

2 TEST FACILITY

The test facility used in the testing to be presented is the Spiral Bevel Gear Test Facility at NASA Glenn Research Center[13-15,19]. A sketch of the facility is

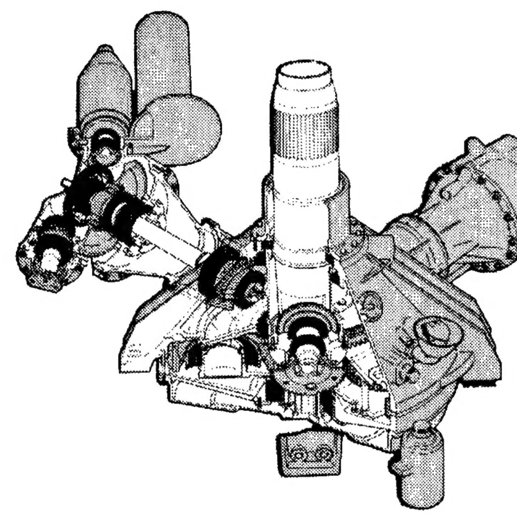

Figure 1.—Main rotor transmission cross-section of the U.S. Army Blackhawk helicopter.

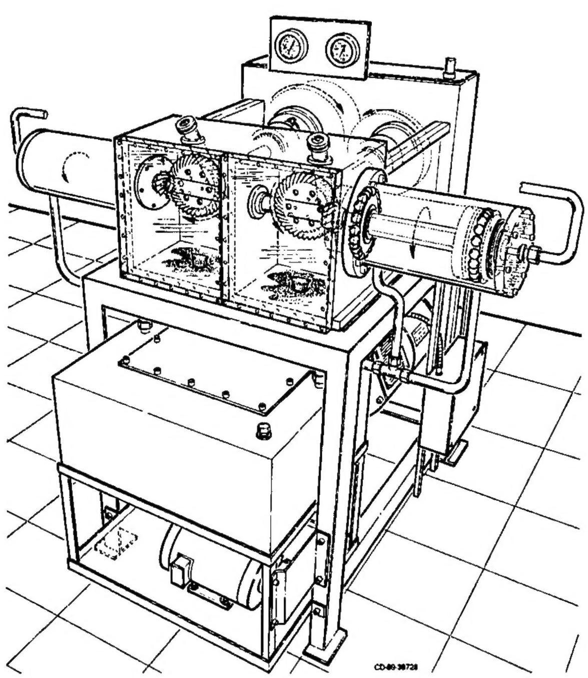

Figure 2.—Spiral bevel test facility sketch.

shown in Fig. 2. An explanation of the operation of the facility is better described in the cross-sectional view showing key features in Fig. 3. The facility operates in a closed-loop arrangement where the load is locked into the loop via a split shaft and a thrust piston that forces a floating helical gear into mesh. The drive motor supplies the facility with rotation and loop losses via v-belts to the axially stationary helical gear. The spiral bevel gears on the test side (left side) operates where the pinion drives the gear in the normal speed reducer mode and the slave side (right side) of the facility acts as a speed increaser, however the concave side of the pinion is always in contact with the convex side of the gear on either side

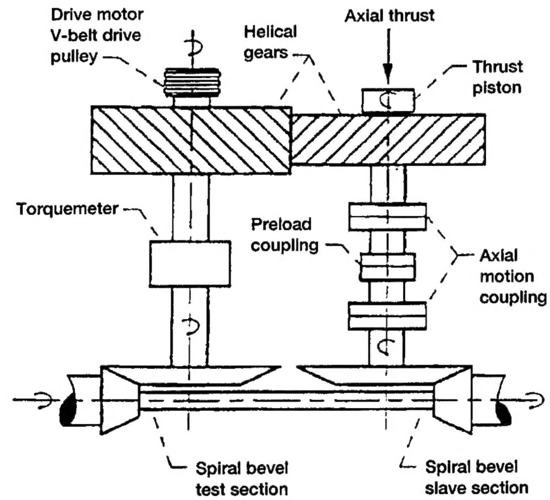

Figure 3.—Cross-sectional view of facility.

Table 1 Facility operational capabilities

Maximum pinion shaft speed, RPM	14400
Pitch line velocity, m/s (ft/min)	44.7 (8803)
Pinion shaft power, kW (hp)	536 (720)
Test section flow rate maximum, cm^3/s (gal/min)	51 (0.8)
Oil inlet temperature (variable), °C (°F)	38 to 189 (100 to 300)
Oil pressure maximum, MPa (psi)	1.38 (200)

Table 2 Basic test component design parameters

Number of teeth pinion / gear	12 / 36
Module, mm (diametral pitch, 1/in.)	4.94 (5.141)
Mean spiral angle, deg.	35
Mean cone distance, mm (in.)	81.1 (3.191)
Face width, mm (in.)	25.4 (1.0)
Shaft angle, deg.	90.0

Table 3 Properties of lubricant used in the test program

Kinematic viscosity, cm^2/sec (cS) at:	
244 K (–20 °F)	2500×10^{-2} (2500)
311 K (100 °F)	31.6×10^{-2} (31.6)
372 K (210 °F)	5.5×10^{-2} (5.5)
477 K (400 °F)	2.0×10^{-2} (2.0)
Flash point, K (°F)	508 (455)
Fire point, K (°F)	533 (500)
Pour point, K (°F)	219 (–65)
Specific gravity	0.8285
Vapor pressure at 311 K (100 °F), torr	0.1
Specific heat at 311 K (100 °F), J/Kg•K, (Btu/lb•°F)	2190 (0.523)

of the test facility. The load and speed of the test side gear output shaft are monitored by an in-the-loop torque and speed sensors. The facility was operated up to 14400 rpm and 354 N•m (3130 in.•lb) pinion speed and torque. The facility operational capabilities and basic gear specific-ations are shown in Tables 1 and 2, respectively.

As mentioned earlier the test gear meshes and facility gear systems are jet feed lubricant and gravity drained back to the reservoir. A single jet lubricated the test and slave side spiral bevel gears. The lubricant supply pressure (flow) and temperature of the lubricant to the jets was controlled during all tests. A turbine engine lubricant was used in all tests. The properties of the lubricant used in this study are shown in Table 3. A

vacuum is applied the gear boxes in the facility to aid in the draining of the lubricant out of the test and facility gear boxes and bearing compartments.

3 TEST HARDWARE INSTRUMENTATION

The testing that will be described in this paper had the pinion, left side (test side), instrumented for the thermal and structural tests. In both tests, data was transferred from the rotating instrumentation via slip rings to either a laboratory computer or tape recorder.

3.1 Thermal Testing

The thermal testing had thermocouples imbedded in the pinion teeth such that the meshing action did not have direct contact with the instrumentation. In studies using this facility[14] different approaches to trying to extract transient temperature data from thin film sensors or thermocouples imbedded in the meshing gear profiles of the gear teeth were not successful. However, important information can be extracted from the bulk temperatures to be described in this report and show the general trends of how the different parts of the pinion teeth thermally operated. The general layout of the gage location is shown in Fig. 4. The thermocouples were located at the half-face width for the tooth top and root gage. For the toe and heel thermocouples the thermocouples were located at the mid point between the concave and convex sides of the teeth and at the mid-tooth height location. A photograph of the test hardware used is shown in Fig. 5.

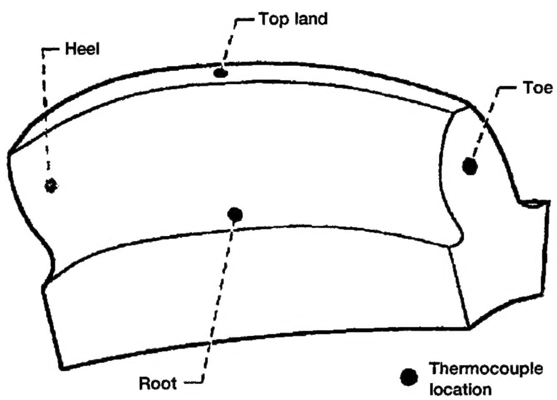

Figure 4.—Thermocouple locations on test pinion.

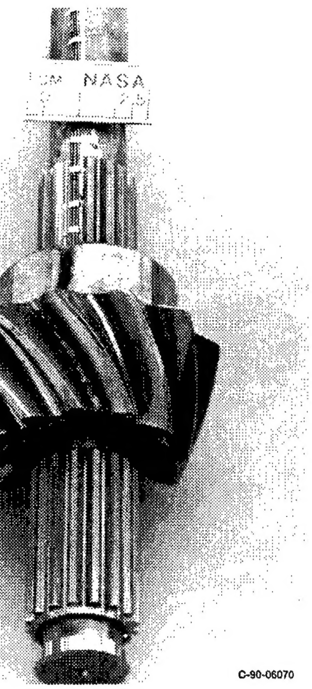

Figure 5.—Photograph of test pinion for thermal tests.

3.2 Structural Testing

The structural testing that was conducted in the test to be described used strain gages in the fillet region[19]. The locations of the strain gages as mounted on the test pinion are shown in Fig. 6 and a photograph of the test

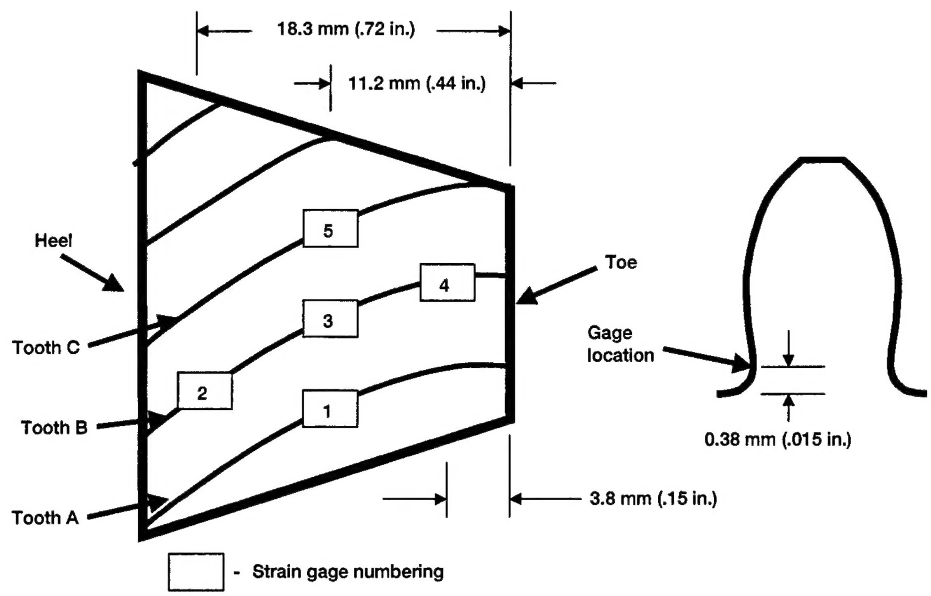

Figure 6.—Strain gage location on the three successive teeth along the pinion fillet.

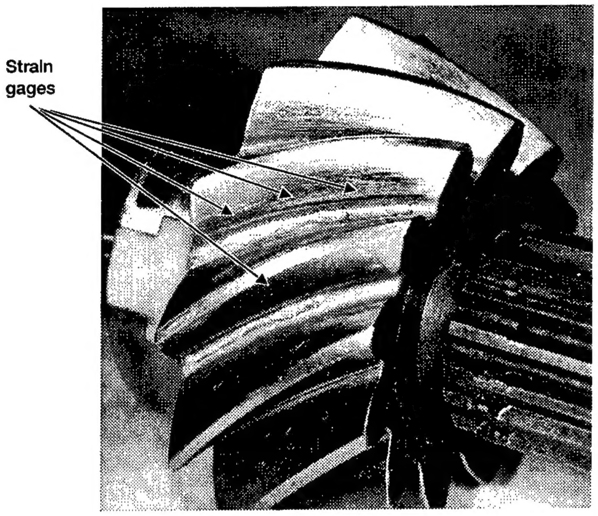

Figure 7.—Strain gaged pinion used in experiments.

hardware is shown in Fig. 7. Testing using these gages was from near static (slow roll) to high speed. Data taken above the slow-roll conditions were time synchronous averaged to reduce the noise from slip rings without using low-pass filtering.

4 THERMAL TESTING RESULTS

The spiral bevel gear meshes in the test and slave side of the test rig were lubricated with a single jet with a fan spray. The fan spray was oriented at a distance from the tooth tip such that the width of the fan was approximately equal to the face width of the gear members. Test data was taken after steady state had been reached. A typical time history of a test is shown in Fig. 8. After about 10 min of operation, steady state conditions were reached. Also from this data, a general trend as seen in all data that was taken. The tooth top was always the highest temperature. This was followed by the toe and root regions that produced similar results at a significantly lower temperature followed by the heel location at the gear teeth.

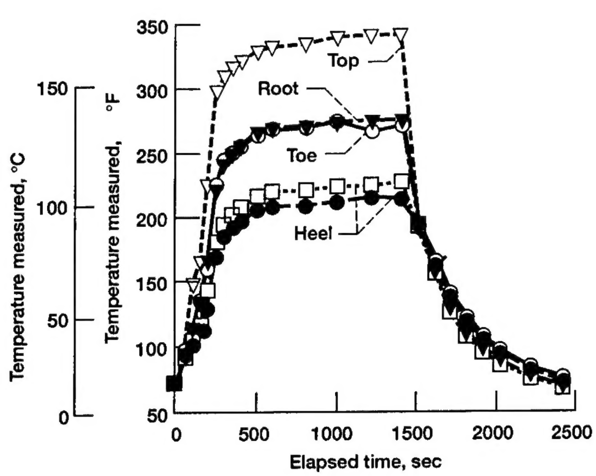

Figure 8.—Typical time history of a test from initiation of a test to steady state conditions.

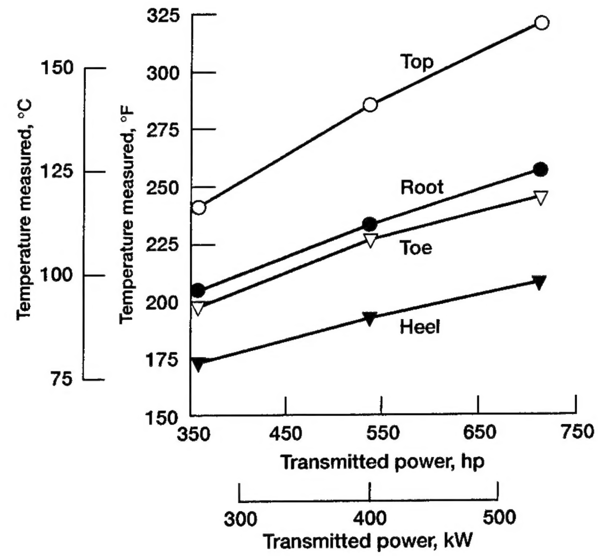

Figure 9.—Effect of load on pinion tooth temperatures at three power levels. All data taken at 38 °C (100 °F) oil inlet temperature and 14 400 rpm pinion shaft speed.

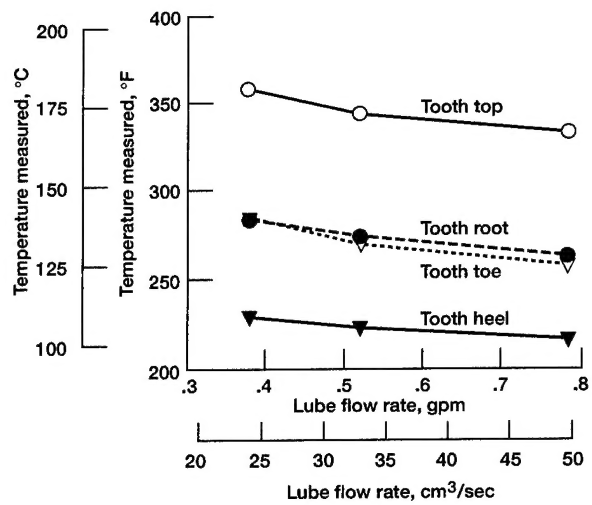

Figure 10.—Effect of lubricant flow rate on temperature.

Thermal effects of several variables as measured will now be presented and discussed. The first effect to be investigated is that of applied load. As shown in Fig. 9, the increase in load had a linear effect on the temperature increase on all four thermocouple locations. The next effect to be investigated was the effect of lubricant jet pressure (or flow rate) of the lubricant to the gear mesh. The effect of flow rate on the steady state temperature is shown in Fig. 10. The flow rate was varied from 24 to 49 cm^3/sec (0.38 to 0.78 gpm) with a slight linear reduction in operating temperature over the range of tests. This flow rate variation was achieved by varying the lubricating jet pressure from ~0.35 to 1.42 MPa (50 to 205 psi). The last effect and a very important aspect to be investigated was the effect of

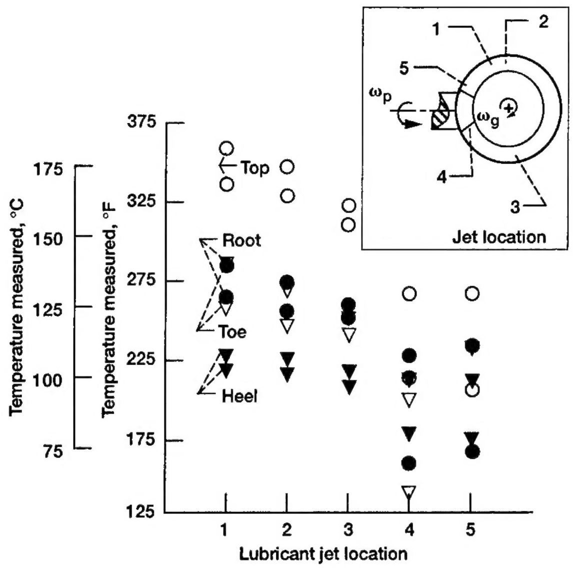

Figure 11.—Effect of lubricant jet placement on pinion temperature for two different flow rates with the load approximately 532 kW (714 hp). Upper symbol is for 24 cm^3/sec (0.38 gpm) and lower symbol is for 49 cm^3/sec (0.77 gpm).

lubricant jet placement on the thermal behavior of the spiral bevel gear mesh. The full load and speed conditions used for this gear mesh were kept nearly constant. Two different lubricating jet pressures were used. The jet was placed at five different locations. The jet locations and results that were attained are shown in Fig. 11. Lubricating away from the into-mesh or out-of-mesh locations created the highest gear tooth temperatures. For the into-mesh or out-of-mesh locations the temperature results were nearly the same. Therefore to make the gear mesh as efficient as possible, the out of mesh jet placement would be the best possible location to orient the jet.

5 STRUCTURAL TESTING RESULTS

The spiral bevel gears used for structural testing had strain gages mounted in the fillet region of the teeth. The locations of the gages, with respect to each other, are shown in Fig. 6. A photograph of the test hardware is shown in Fig. 7. The data from the rotating instrumented pinion was transferred to signal conditioners (the rest of the bridge circuit) via a slip ring. The signals were then output to a tape recorder for post processing. No low-pass filtering of the signals was made. The data for the strain gage tests were taken from static (slow-roll) to 14400 rpm and from light to high load. A summary of the test results will now be presented.

First an example of the slow-roll data is shown in Fig. 12. As can be seen from the data, the three midface fillet gages produced the highest positive bending stress, however between the gages had slightly different amplitudes (a trend seen in all data taken). The gages across the one tooth with a heel and toe gage, had the midface being the highest stress with the heel the next highest and the toe position producing the lowest.

The dynamic data taken will be now discussed. The recorded dynamic data was downloaded to a personal computer via an analog to digital converter board and then time synchronous averaged for 100 rotations (typical for higher shaft speeds) of the gear shaft. An example of the dynamic data from the three midface fillet gages is shown in Fig. 13. As can be seen the data taken by the three gages were fairly consistent with

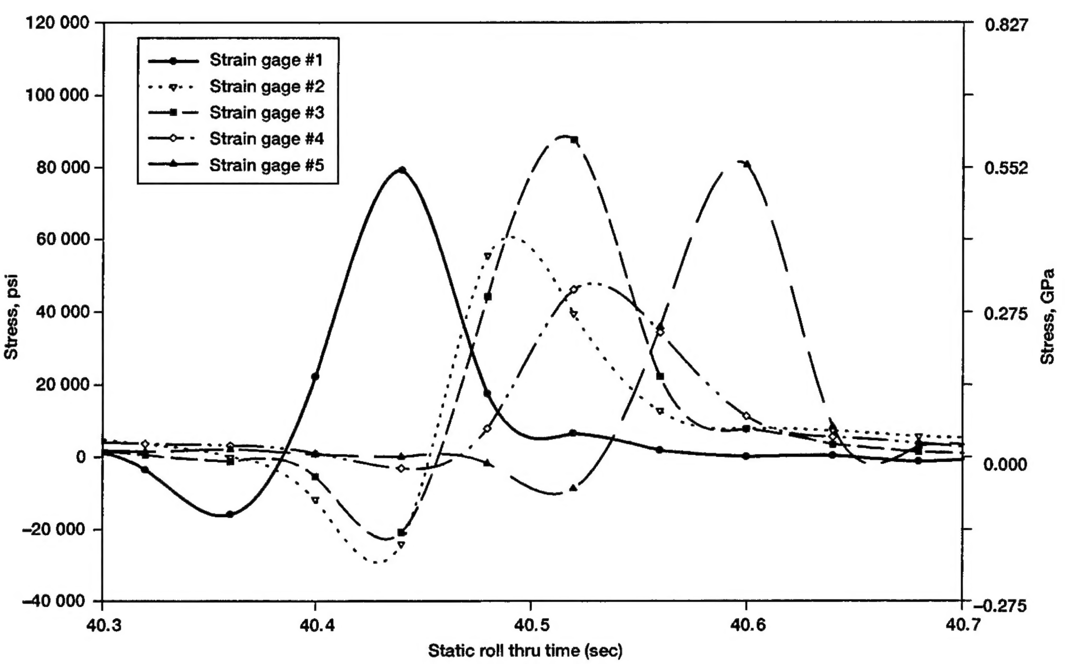

Figure 12.—Example of slow-roll bending stress at 885.8 N•m (7840 in.•lb) gear shaft torque.

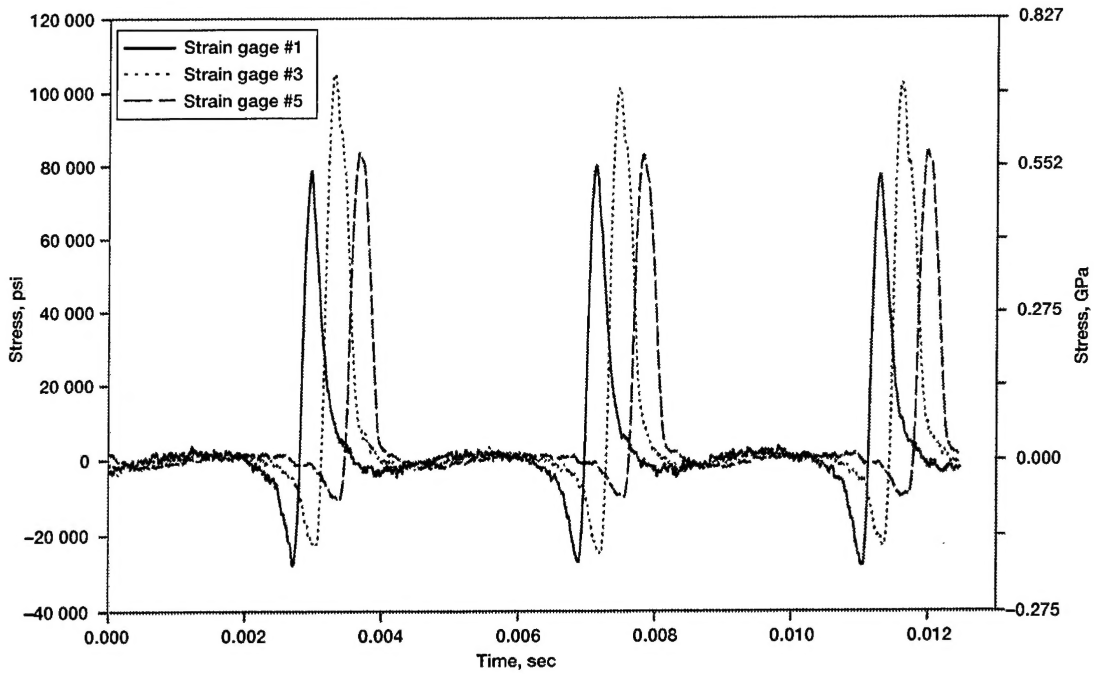

Figure 13.—Dynamic data for one rotation of the gear shaft at 1073 N•m (9500 in.•lb) torque and 14 400 rpm pinion shaft speed.

respect to the relative stress differential between the gages even though they were all located at the same midface position. The reason that this occurs is due to the gage placement in the fillet region. Even though the gages were only 0.38 mm (0.015 in.) active gage length, the fillet radius of the pinion is not much larger than the gage active length. Also, based on prior finite element analysis of gears, the strain field varies substantially around the fillet. Therefore the gage averages the strain field over the gage length and the placement location. Gage location too high on the fillet or too low will result in a slightly different strain (stress) result. As can be seen in Fig. 13, the three fillet gage results indicate that gage #1 was located more toward the root than either of the other two midface fillet gages as this gage had the largest compressive stress (an effect that is seen by further locating the gages in the root region), and gage #5 had the lowest compressive strain (stress) that meant the gage was mounted slightly higher up the fillet thus resulting in a lower compressive and tensile components.

The last comparison to be made is the midface strain gage output as a function of speed and applied load. The results are shown in Fig. 14. The maximum stress value was found from the dynamic data by scanning the time synchronous averaged data at each of the conditions shown. The gear shaft torque was varied approximately from 452 N•m (4000 in.•lb) to 1186 N•m (10500 in.•lb) at three different levels of rotational speed. As can be seen from the data, the rotational speed of the pinion had little effect on the bending stress that was measured. This meant that there were minimal dynamic effects at the conditions measured.

6 CONCLUSIONS

Based on the results attained in this study the following basic conclusions can be made:

(1) All parameters that were varied in the thermal studies had an effect on the steady state tooth temperatures. Tooth temperature was the highest at the tooth top (midface position) with the highest temperature in excess of 188 °C (370 °F). The temperature measured at the root and toe were similar over all the test conditions measured and the temperature was the lowest at the heel position.

(2) Both into-mesh and out-of-mesh lubricant jet placement produced the lowest tooth temperatures. However the out-of-mesh position would be the better choice for the optimum position as the lubricant would not need to be pumped out of the meshing zone in that position.

(3) The stress measurements indicated that the pinion rotational speed had a minimal effect on the maximum stress from slow-roll to 14400 rpm. Therefore no dynamic effect was evident over the rotational speeds tested.

(4) The midface position produced the highest fillet stress followed by the heel then the toe locations.

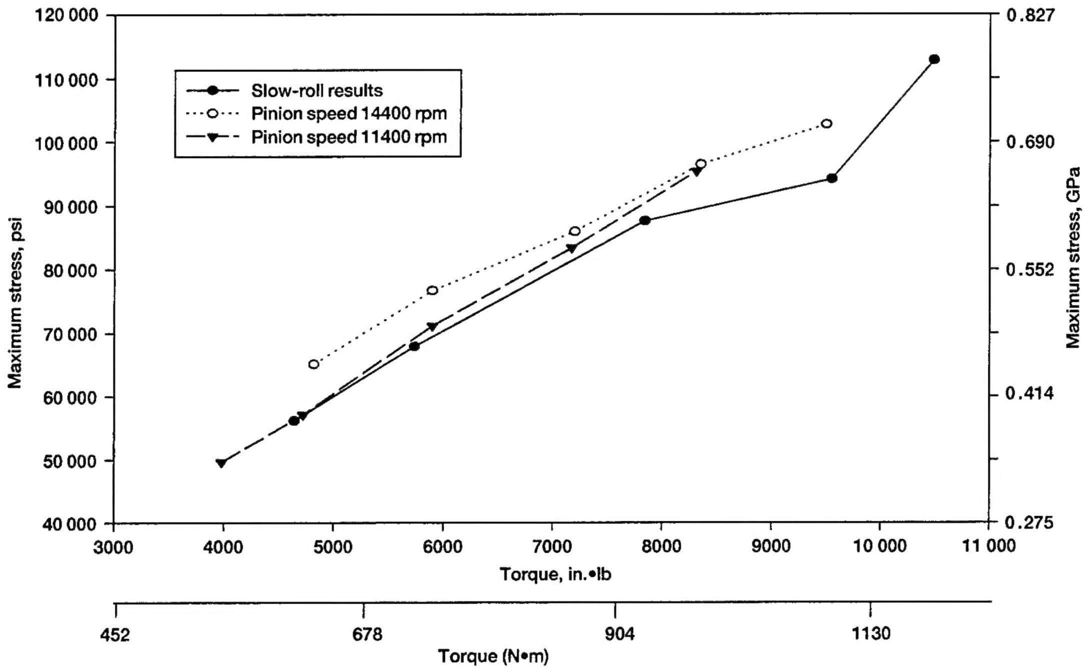

Figure 14.—Comparison of slow roll data to that taken at two pinion shaft speeds. Applied torque was measured at the gear shaft.

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